

AD-A016 925

INVESTIGATION OF ACCELERATED LIFE PREDICTION
TECHNIQUES

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Ohio State University

Prepared for:

Army Air Mobility Research and Development Laboratory

October 1975

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USAAMRDL-TR-75-38



**Department of Mechanical Engineering
The Ohio State University Research Foundation
Columbus, Ohio 43210**

October 1975

Final Report for Period 1 March 1974 - 28 February 1975

Approved for public release;
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Prepared for

EUSTIS DIRECTORATE

U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
Fort Eustis, Va. 23604

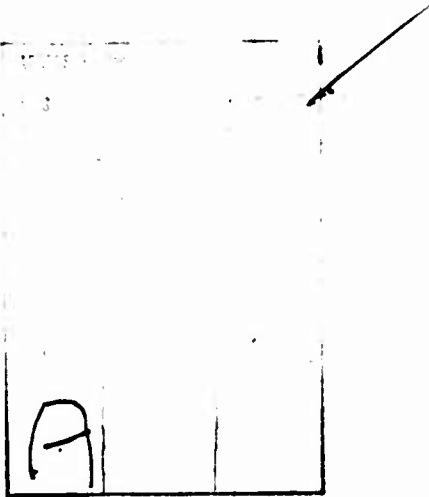
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Three failure prediction models (adhesive/abrasive, zero-wear, and linear cumulative damage) were selected as being potentially useful for predicting the time to failure caused by wear and fretting-wear. The parameters necessary for predicting wear and fretting-wear failures were identified, and a testing program was designed to provide data to validate these models when applied to the prediction of wear and fretting-wear failures. A machine suitable for conducting component wear and fretting-wear life testing was designed. The accelerated life tests were not conducted as part of the contract.

The conclusions contained herein are concurred in by this Directorate.

The technical monitors for this contract were Mr. Howard M. Bratt and Mr. Gary R. Newport, Military Operations Technology Division.



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REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER USAAMRDL-TR-75-38	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) INVESTIGATION OF ACCELERATED LIFE PREDICTION TECHNIQUES		5. TYPE OF REPORT & PERIOD COVERED Final Report 1 Mar 1974 thru 28 Feb 1975
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Jack A. Collins Ben Tarver Hagan, Jr.		8. CONTRACT OR GRANT NUMBER(s) Contract DAAJ02-74-C-0033
9. PERFORMING ORGANIZATION NAME AND ADDRESS Department of Mechanical Engineering The Ohio State University Research Foundation Columbus, Ohio 43210		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS 62203A 1F262203AH86 03 003 EK
11. CONTROLLING OFFICE NAME AND ADDRESS Eustis Directorate U. S. Army Air Mobility R&D Laboratory Fort Eustis, Virginia 23604		12. REPORT DATE October 1975
		13. NUMBER OF PAGES 59
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION DOWNGRADING SCHEDULE
16. DISTRIBUTION STATEMENT (of this Report) Approved for public release; distribution unlimited.		
17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)		
18. SUPPLEMENTARY NOTES		
19. KEY WORDS (Continue on reverse side if necessary and identify by block number) Accelerated testing Life Failure Predictions Fretting Testing Machines Helicopters Wear		
20. ABSTRACT (Continue on reverse side if necessary and identify by block number) The objectives of the work reported were to define potentially useful failure prediction models for the wear and fretting-wear modes of failure, to define the parameters of primary importance, to incorporate the concept of accelerated testing both in the prediction models and in the design of a testing program, and to design special testing machines capable of both real-time and accelerated-wear and fretting-wear tests using a UH-1H helicopter cyclic servo support bearing as the test specimen. This		

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20. Continued.

particular component was selected because it was a relatively simple example of an actual component in which both wear and fretting-wear failure modes had been regularly observed in the field.

The four objectives described above were accomplished in the study reported here. An extensive bibliography of reference material was compiled in the process and is included at the end of this report. The design of a special testing machine for accelerated wear and fretting-wear tests was completed, with a full set of engineering drawings. The actual construction of the testing machine and performance of the experimental testing program remain to be accomplished in a future program.

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PREFACE

This investigation was conducted under Contract DAAJ02-74-C-0033, administered by Eustis Directorate of the U. S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia. The contracting officer's technical representative monitoring this contract was Mr. Gary Newport. The authors wish to express appreciation to Mr. Gary Newport and Mr. Howard Bratt of the Eustis Directorate for providing essential information and documentation throughout the contract period.

Supporting efforts by Ohio State University graduate students Max Herschler, Harvey Lyons, and Imtiyaz Syed are recognized.

The period of time covered by this investigation was from March 1, 1974, through February 28, 1975.

TABLE OF CONTENTS

	<u>Page</u>
PREFACE	3
LIST OF ILLUSTRATIONS	6
LIST OF TABLES	7
INTRODUCTION	8
PREDICTION MODELS AND ACCELERATED TESTING FOR WEAR AND FRETTING-WEAR FAILURE MODES	10
PROPOSED TESTING PROGRAM	14
TESTING MACHINE REQUIREMENTS	20
DESIGN OF FRETTING-WEAR TESTING MACHINE	24
Specimen Configurations	24
Method Selected for Producing Forces and Motions	24
Force Control and Measurement	25
Two Motion Ranges	26
Motion Measurement	26
Torque Measurement	27
Balancing of Moving Parts	28
Temperature Measurement	29
Facilities for Cooling the Specimen	29
Damage Measurement and Failure Detection	30
SUMMARY AND CONCLUSIONS	31
REFERENCES	33
SELECTED BIBLIOGRAPHY	34
LIST OF SYMBOLS	57

LIST OF ILLUSTRATIONS

<u>Figure</u>		<u>Page</u>
1	Example of Accelerated Life Testing Using the Linear Cumulative Damage Model	12
2	Schematic View of Cyclic Servo Support Bearing	21

LIST OF TABLES

<u>Table</u>		<u>Page</u>
1	Proposed Static and Dynamic Baseline Control Conditions for Use in Testing UH-1H Cyclic Servo Support Bearing Specimens	14
2	Proposed Preliminary Series for Verifying the Validity of the Accelerated Test Model and Wear Prediction Equations Using UH-1H Cyclic Servo Support Bearings as Test Specimens	16
3	Accelerated Wear Testing Program Using Operational Load Amplitude as the Stressor	17
4	Accelerated Wear Testing Program Using Operational Frequency of Angular Motion as the Stressor	18
5	Ranges of Important Parameters Established as Requirements for the Wear/Fretting-Wear Testing Machine	22

INTRODUCTION

Wear and fretting-wear are extremely important mechanical failure modes with a very high incidence of occurrence in all types of mechanical equipment. Wear, usually associated with sliding surfaces in contact, results in failure when the changes in dimensions of the mating parts, due to the gradual removal of material from the contacting surfaces, become large enough that the parts are no longer able to properly perform their design function. Fretting-wear failure occurs when the changes in dimensions of the mating parts, due to the presence of fretting action, become large enough to interfere with proper design functions, or large enough to produce geometrical stress concentration of such magnitude that failure ensues due to excessive local stress levels. It is important to note the distinction between wear, which is caused by unidirectional sliding or large amplitude cyclic sliding between two mating parts, and fretting-wear which takes place at the interface between any two solid bodies pressed together by a normal force and subjected to small-amplitude cyclic relative motion with respect to each other. Typically, in the case of fretting-wear the wear debris is trapped between the contacting surfaces which may move only a few thousandths of an inch with respect to each other. In the case of wear, the amplitudes of relative motion are large enough to spill the debris from the wearing interface.

In some instances, wear or fretting-wear failure may simply result from a loss of proper fit or a change in dimension which requires the replacement of a worn part. In other cases, wear or fretting-wear failure may result in loss of function, seizure, or loss in control of a critical system to produce a catastrophic failure. Of particular interest to the investigation reported here are the results of a study of more than 500 documented mechanical failures in U. S. Army helicopter subsystems and components.[1]¹ While approximately 40 different failure modes were identified in the study, it was found that more than half of all failures were directly attributable to just two of these failure modes, namely, wear and fretting-wear.

Because of the observed high incidence of failure by wear and fretting-wear, the long range objectives of mitigating and/or preventing wear-related failures, especially in critical components, were clearly indicated. The work reported here represents completion of the first phase in an effort to accomplish the long-term objectives. The primary objectives of the work reported here were: (1) to define potentially useful failure prediction models for the wear and fretting-wear modes of failure, (2) to define the parameters of primary importance, (3) to incorporate the concept of accelerated testing both in the prediction models and in the design of a testing program, and (4) to design special testing machines capable of both real-time and accelerated wear and

¹ Numbers in brackets designate references cited at end of report.

fretting-wear tests using a UH-1H helicopter cyclic servo support bearing as the test specimen. This particular component was selected because it was a relatively simple example of an actual component in which both wear and fretting-wear failure modes had been regularly observed in the field.

The four objectives described above were accomplished. An extensive bibliography of reference material was compiled in the process, and is included at the end of this report. The design of a special testing machine for accelerated wear and fretting-wear tests was completed, with a full set of engineering drawings. The actual construction of the testing machine and performance of the experimental testing program remain to be accomplished in a future program.

PREDICTION MODELS AND ACCELERATED TESTING FOR WEAR AND FRETTING-WEAR FAILURE MODES

From the study of potentially useful failure prediction models for wear and fretting-wear failures which are valid for both real-time and accelerated tests, it has been concluded that three prediction models should be experimentally investigated. These are the adhesive/abrasive model [2,3], the zero-wear model [4], and the linear cumulative damage model [5]. Since the primary independent variables are thought to be the same, except for magnitude, for both wear and fretting-wear, these three models should be applicable for either the wear failure mode or the fretting-wear failure mode if the constants are properly determined by appropriate experiments.

The adhesive/abrasive wear model is defined by [2,3]

$$\left. \begin{array}{ll} d_w = k_w p_m L_s & \text{for } p_m < \sigma_{yp} \\ \text{unstable galling and seizure} & \text{for } p_m \geq \sigma_{yp} \end{array} \right\} \quad (1)$$

where d_w is mean depth of the wear, p_m is mean nominal contact pressure, L_s is distance of sliding, and k_w is an adhesive/abrasive wear parameter to be determined experimentally. While adhesive wear and abrasive wear have each been investigated over a limited range of conditions and materials, and successfully correlated to a model of the general type given in equation (1), general engineering design information has not been developed and reliable values of the parameter k_w have not been established for realistic engineering design situations. Also, the concept of using experimental evaluation of the parameter k_w over the full range of adhesive and abrasive wear behavior has not been reported in the literature. Thus, while the validity of equation (1) has been established for a variety of specific conditions, much additional experimentation is required to validate the model for general engineering use. Further, no work has been reported in the literature with respect to the use of equation (1) for modeling the fretting-wear phenomenon.

The term "zero wear" is defined to be wear of such small magnitude that the wear depth is the order of one-half the peak-to-peak surface finish dimension. The zero wear model is defined by [4]

$$N = 2 \times 10^3 \left[\frac{\gamma_r \tau_{yp}}{\tau_{max}} \right]^9 \quad (2)$$

where N is the maximum number of passes for zero wear, τ_{yp} is the shear yield point strength at the surface of the softer material, τ_{max} is the maximum shearing stress in the vicinity of the surface due to both normal and friction forces, and γ_r is a constant to be experimentally determined. The zero-wear model has been validated for a wide range of

wear conditions and materials and a useful body of engineering data is available for prediction of zero-wear lives under many wear conditions [4]. However, no work has been reported in the literature with respect to the use of equation (2) for modeling the fretting-wear phenomenon.

The generalized linear cumulative damage model [5] requires a minimum of specific information about the physical process involved and is most readily adaptable to accelerated testing. However, the validity of this model has not yet been shown. To use the linear cumulative damage model, a test accelerating factor, such as normal load P , is selected as the "stressor". The term "stressor" is used to designate any generalized stress-like quantity that may be varied to accelerate a life test. For example, temperature, load, or environment might be used as stressors in wear or fretting-wear tests. The generalized linear cumulative damage relationship implies that, for a specified failure mode, if the component life is L_1 at stressor level S_1 , and the life is L_2 at stressor level S_2 , then if the component is subjected to operation at stressor level S_1 for a time αL_1 , it will have a remaining life at stressor S_2 of βL_2 , where

$$\alpha + \beta = 1 \quad (3)$$

The validity of this relationship for the wear mode of failure remains to be established. Experimental verification would be indicated if two groups of specimens were tested as follows: Group I specimens would be tested at stressor level S_1 for a time t_1 followed by testing at stressor level S_2 , with failure observed after time t_2 at level S_2 . Group II specimens then would first be tested at stressor level S_2 for a time t_2 followed by testing at stressor level S_1 , with failure observed after time t_3 at level S_1 . If $t_3 = t_1$, the hypothesis would be verified for this combination of conditions. Testing over a suitable range of stressor levels in various combinations would verify the model. It has been shown that for many types of failure this linear model seems to hold with reasonable accuracy.

If the model were verified, then accelerated life testing, using normal load P as the stressor, would consist of running specimens first at a high load level, P_{hi} , until failure occurs. Such a failure point is shown as "A" in Figure 1. Next, specimens would be run at normal operating load level P_{op} for a fraction α of the operating life time and then tested to failure at P_{hi} . This failure point is shown as "C" in Figure 1. Connecting the two points A and C with a straight line, and extrapolating the line to the horizontal axis, gives the life prediction for full operation at P_{op} only. This predicted life is indicated as point "B" in Figure 1. Since most of the testing time would be at the high load level, the total test time would be reduced significantly.

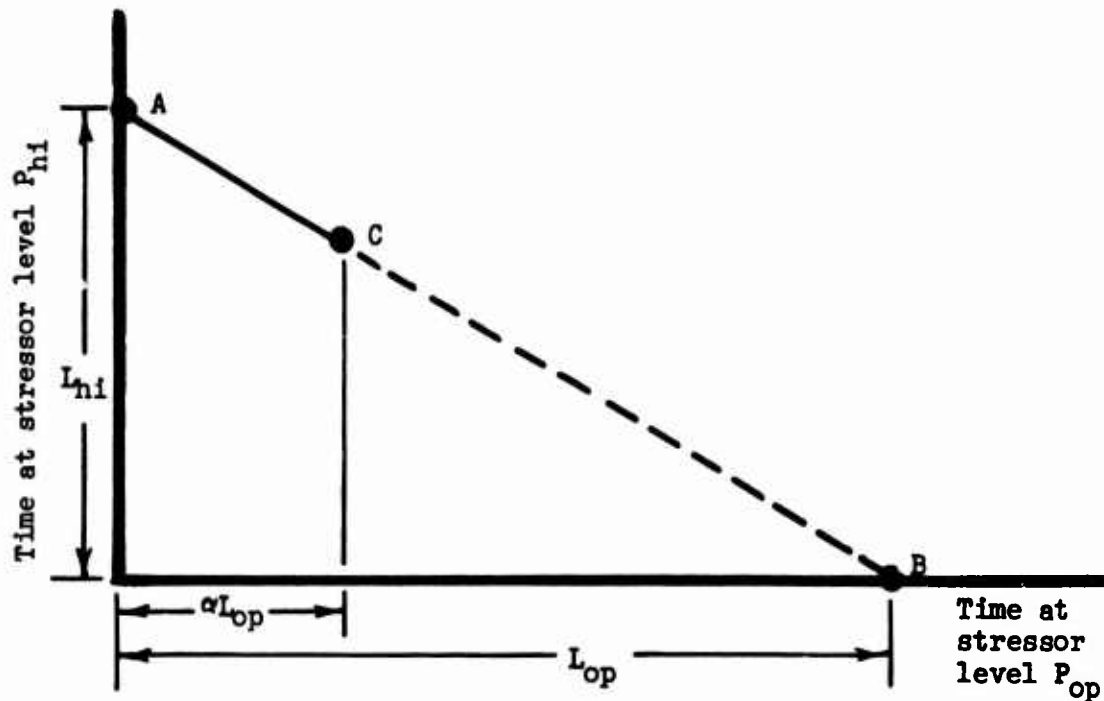


Figure 1. Example of Accelerated Life Testing Using the Linear Cumulative Damage Model.

For example, if life L_{hi} at the higher load level were 10 percent of life L_{op} at the operating load level, and test time for point C at the operating stressor level were arbitrarily selected as 10 percent of L_{op} , the total testing time to produce both of the data points A and C would be $0.29 L_{op}$. This would include $0.10 L_{op}$ at P_{hi} for point A plus $0.10 L_{op}$ at P_{op} followed by $0.90(0.10 L_{op})$ at P_{hi} for point C. Thus, the test time would have been accelerated by approximately a factor of four, and since the tests at point A could probably have been run simultaneously with the tests at point C, the effective test-time acceleration factor would have exceeded five.

An extensive literature survey, augmented by independent research, has indicated that there are twelve parameters of primary interest in studying wear or fretting-wear failures. These twelve parameters have been distilled from a list of over fifty parameters which have been postulated to have some influence on the wear processes. From experimental results reported in the literature, based primarily on single parameter studies, the primary variables influencing wear and fretting-wear failures are:

- (1) Amplitude of normal static or cyclic force between contacting surfaces.

- (2) Frequency of normal static or cyclic force.
- (3) Amplitude of relative cyclic motion.
- (4) Frequency of relative cyclic motion.
- (5) Sliding distance and velocity.
- (6) Coefficient of friction.
- (7) Maximum shearing stress in vicinity of surface, including frictional influence.
- (8) Mean depth and volume of wear track.
- (9) Tensile and shear yield strength of surface material.
- (10) Hardness of surface material.
- (11) Characteristic temperature.
- (12) Environment, including lubricant or contaminant.

In the design of a testing machine for wear and fretting-wear failure evaluation, it is necessary to control and/or measure each of these twelve parameters. In the design and execution of an experimental program to test the three models defined in equations (1), (2) and (3), it is necessary to control, measure and record these twelve important parameters, allowing controlled changes only in the stressors, as desired. The following three chapters describe in detail a proposed testing program, the requirements for a testing machine capable of properly executing the testing program, and the detailed design of such a testing machine.

PROPOSED TESTING PROGRAM

In designing the testing program proposed below, it was desired to carefully control and/or measure all important parameters in the wear and fretting-wear experiments to provide data both for determining the constants and for testing the validity of the three models described in equations (1), (2) and (3). At the same time, it was desired to use a currently troublesome helicopter component as a test specimen so that the laboratory results might be directly applicable to the solution of a specific problem area. Additional criteria for selection of a suitable test specimen included the need for simplicity, compactness, economy, ease of controlling and measuring forces and motions, and potential for application of coatings, platings, and/or lubricants to the wearing interface. The UH-1H cyclic servo support bearing was selected as a component which met all of these criteria relatively well. This component, consisting of a double-truncated sphere supported by two rings with spherical mating surfaces, is shown schematically in Figure 2. The spherical element has a cylindrical hole through it for securing it to the servo actuator rod. While this component became the specimen of choice in designing the testing machine, the machine was designed so that many other specimen configurations could also be easily accommodated.

From assessment of the operational loads, motions and other parameters associated with the cyclic servo support bearing in service, two sets of laboratory baseline control conditions were established to provide realistic but practical comparison standards. These control conditions, designated "static" and "dynamic", are defined in Table 1.

TABLE 1. PROPOSED STATIC AND DYNAMIC BASELINE CONTROL CONDITIONS FOR USE IN TESTING UH-1H CYCLIC SERVO SUPPORT BEARING SPECIMENS		
Parameter Description	Parameter Value	
	Static	Dynamic
Magnitude of axial preload force, lb	960	260
Amplitude (half the peak-to-peak) of cyclic axial force about the mean, lb	0	700
Frequency of cyclic axial force, cycles/min	0	600
Peak-to-peak magnitude of maximum relative cyclic motion, in.	0.030	0.030
Frequency of cyclic motion (with motion midrange in phase with force peak), cycles/min	600	600

TABLE 1 - Continued		
Parameter Description	Parameter Value	
	Static	Dynamic
Defined useful life, hr	600	600
Ambient environment	Laboratory Air	Laboratory Air
Lubricant	None	None

In conducting tests using either these baseline control conditions or accelerated test conditions, it is necessary to carefully determine the following dependent variables for all tests:

- (a) Profile of relative sliding velocity.
- (b) Sliding distance.
- (c) Torque to move bearing element relative to supporting races.
- (d) Nominal coefficient of friction.
- (e) Maximum shearing stress in vicinity of contacting surfaces.
- (f) Tensile and shear yield strength of surface material.
- (g) Hardness of surface material.
- (h) Characteristic temperature near contacting surfaces.
- (i) Characteristic depth and volume of wear track.

The design of the testing machine and the experimental data collection procedure must insure that all of these parameters can be evaluated.

As a first phase of the testing program, it is proposed that a preliminary test series be conducted to assure the feasibility of using the linear cumulative damage model of equation (3), as well as to obtain preliminary values for the constants in equations (1) and (2) for the adhesive/abrasive wear model and the zero wear model. It is proposed to use

approximately 30 specimens for this preliminary test series, which is described in detail in Table 2.

TABLE 2. PROPOSED PRELIMINARY SERIES FOR VERIFYING THE VALIDITY OF THE ACCELERATED TEST MODEL AND WEAR PREDICTION EQUATIONS USING UH-1H CYCLIC SERVO SUPPORT BEARINGS AS TEST SPECIMENS				
Test No.	Test Conditions	No. of Specimens	No. of Data Points ¹	Estimated ² Test Duration, days
1	Static baseline as defined in Table 1 until failure (L_{Ps})	4	8	L_{Ps} = 25 days continuous running (600 hr)
2	Static baseline <u>except</u> static force of 4800 lb, until failure	4	8	5
3	Static baseline <u>except</u> static force of 960 lb for $0.25 L_{Ps}$ followed by static force of 4800 lb, until failure ($T_{0.25}$)	4	8	10
4	Static baseline <u>except</u> static force of 960 lb for $0.75 L_{Ps}$ followed by static force of 4800 lb, until failure ($T_{0.75}$)	4	8	20
5	Static baseline <u>except</u> static force of 4800 lb for $T_{0.25}$ followed by static force of 960 lb, until failure	4	8	10
6	Static baseline <u>except</u> static force of 4800 lb for $T_{0.75}$ followed by static force of 960 lb, until failure	4	8	20
7	Dynamic baseline <u>except</u> axial preload of 1300 lb with cyclic amplitude of 3500 lb, until failure	4	8	5

TABLE 2 - Continued

- ¹ The two ring-ball interfaces provide two separate data points for each specimen tested.
- ² Assumes that two double-test-bed machines of the type described in later sections of this report are both running continuously for this period of time to produce data for four specimens.

If these preliminary tests yield promising results, an extended testing program, involving six test series, is proposed. The extended testing program is in Tables 3 and 4. First, using the axial load as the accelerating parameter (stresser), the two-step test series of Table 3 is proposed. Baseline control conditions tabulated in Table 1 would prevail for all tests, except that axial load P would be assigned selected values as indicated.

TABLE 3. ACCELERATED WEAR TESTING PROGRAM USING OPERATIONAL LOAD AMPLITUDE AS THE STRESSOR

Series (a): Step I		Series (a): Step II	
Load Amplitude and Frequency	Duration of Loading	Load Amplitude and Frequency	Duration of Loading
P (static)	L_{ps} (failure)	--	--
P @ 600 cpm	L_p (failure)	--	--
P " " "	$0.25 L_p$	5 P @ 600 cpm till failure ($t_{0.25}$)	
P " " "	$0.50 L_p$	5P " " " " ($t_{0.50}$)	
P " " "	$0.75 L_p$	5P " " " " ($t_{0.75}$)	
5P @ 600 cpm	$t_{0.25}$	P @ 600 cpm	till failure
5P " " "	$t_{0.50}$	P " " "	" "
5P " " "	$t_{0.75}$	P " " "	" "

TABLE 3 - Continued			
Series (a): Step I		Series (a): Step II	
Load Amplitude and Frequency	Duration of Loading	Load Amplitude and Frequency	Duration of Loading
5P @ 600 cpm	till failure	--	--
5P (static)	till failure	--	--
Series (b) - Identical to series (a), except using 2.5 P instead of 5.0 P.			
Series (c) - Identical to series (a), except using 1.2 P instead of 5.0 P.			

Using frequency of cyclic motion as the stressor, the two-step test series of Table 4 is proposed. Again, the baseline control conditions of Table 1 would prevail for all tests except that the cyclic motion frequency f_θ would be assigned selected values as indicated.

TABLE 4. ACCELERATED WEAR TESTING PROGRAM USING OPERATIONAL FREQUENCY OF ANGULAR MOTION AS THE STRESSOR			
Series (d): Step I		Series (e): Step II	
Oscillating Motion Frequency	Duration of Oscillating Motion	Oscillating Motion Frequency	Duration of Oscillating Motion
f_θ	till failure ¹ $L_{\theta s}$	--	--
f_θ	till failure ² L_θ	--	--
f_θ	0.25 L_θ	5 f_θ	till failure ($t'_{0.25}$)
f_θ	0.50 L_θ	5 f_θ	till failure ($t'_{0.50}$)
f_θ	0.75 L_θ	5 f_θ	" " ($t'_{0.75}$)

TABLE 4 - Continued			
Series (d): Step I		Series (e): Step II	
Oscillating Motion Frequency	Duration of Oscillating Motion	Oscillating Motion Frequency	Duration of Oscillating Motion
$5f_0$	$t'_{0.25}$	f_0	till failure
$5f_0$	$t'_{0.50}$	f_0	" "
$5f_0$	$t'_{0.75}$	f_0	" "
$5f_0$	till static failure	--	--
$5f_0$	till cyclic failure	--	--
Series (e) - Identical to series (d), except using $10 f_0$ instead of $5.0 f_0$.			
Series (f) - Identical to series (d), except using $2.5 f_0$ instead of $5.0 f_0$.			
¹ Under static application of P. This is same data as first item of series (a).			
² Under cyclic application of P @ 600 CPM. This is same data as second item of series (a).			

It is proposed to first utilize four specimens in each test series specified. This would involve ten conditions for each of six series for a total of 240 test specimens. Final statistically significant data would require a minimum of four additional specimens at each of three sets of conditions in at least 3 of the series above for a total of 36 additional specimens. Allowing for approximately 10 percent spoiled runs, the overall program would therefore require approximately 300 cyclic servo support bearings to be used as specimens. It is estimated that if two double-test-bed machines were available for continuous duty, the overall testing program, as proposed, would require a period of two to three years to complete. The long-range potential improvement in wear and fretting-wear failure experience is thought to justify such an investment of time and effort.

TESTING MACHINE REQUIREMENTS

To meet the criteria established for the testing program proposed in the preceding chapter, and to provide the additional flexibility for testing different material combinations, finishes, platings, coatings, lubricants, and geometrical configurations in a wear or fretting-wear environment, it was necessary to establish proper characteristic ranges for all of the important parameters involved. In the case of both wear and fretting-wear, normal force and sliding motion always exist between the contacting surfaces. Any externally applied force must have a component normal to the contacting surfaces in order to bring the surfaces into contact. Such a force may remain constant or it may fluctuate with time, often in a periodic pattern about some mean value. The relative sliding motion between the contacting areas may be either unidirectional or reciprocating for the case of wear, but for fretting-wear it is characteristically reciprocating motion. For some cases of adhesive wear or fretting-wear, a net macroscopic tangential sliding motion may not be necessary since simply pressing together and pulling apart the surfaces by a fluctuating normal force may be sufficient to produce transverse elastic strains which result in local sliding at the interface. In any case, a cyclic normal force and an independent normal preload force combined with a reciprocating motion was thought to provide a very flexible arrangement for producing realistic wear and fretting-wear conditions in the laboratory.

Fretting-wear and wear may occur with a wide variety of geometries ranging from complex gear-teeth and cam surfaces to relatively simple "flat" surfaces. The size of affected parts may be exceedingly small or very large. For practical reasons, relatively simple, frequently encountered geometries manufactured in reasonably small sizes and requiring only modest power consumption were judged suitable for testing consideration. Evaluation of these considerations resulted in the decision to include, as a minimum, provisions for testing both spherical and cylindrical configurations. These two basic geometries are frequently encountered in bearing applications. The cylindrical geometry is the simpler of the two to evaluate, but a large number of bearings involve spherical elements. It was also decided that the test specimen should be restricted to a size which could be contained within an envelope of approximately 3 inches x 3 inches x 3 inches, be capable of being disassembled for inspection, and be compatible with reciprocating motion operation. For the initial testing program, a component known to have experienced wear and/or fretting-wear damage in the field was preferred. To provide an opportunity for comparison of real-time and accelerated testing results with field results, the component actually proposed as a test specimen was the cyclic servo support bearing used on UH-1H Army helicopters. The bearing assembly, shown schematically in Figure 2, consisted of a spherical element 2.154 inches to 2.156 inches in diameter that oscillated in a pair of separable, supporting bushings.

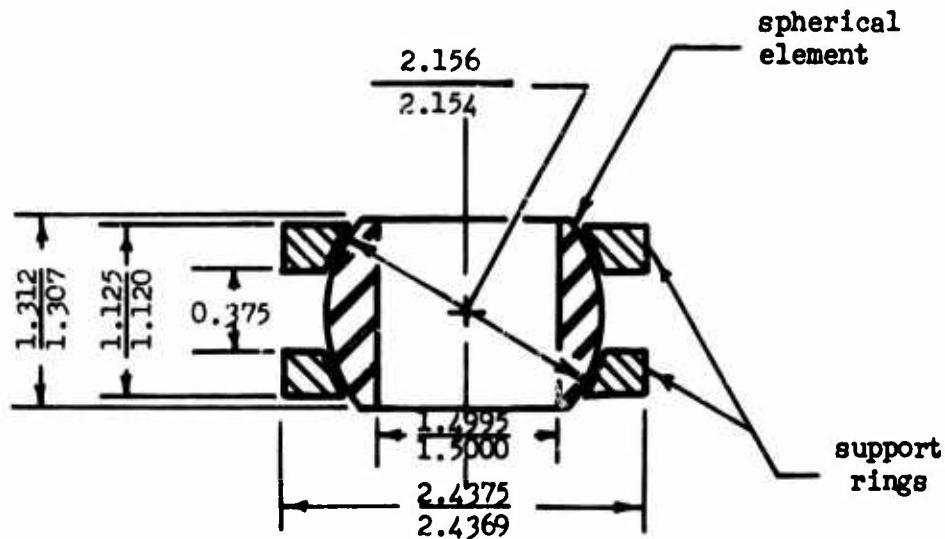


Figure 2. Schematic View of Cyclic Servo Support Bearing.

From operational load and motion information and a knowledge of the fretting-wear and wear phenomena, the standard or baseline conditions of Table 1 were established for this test specimen and are thought to be suitable for many other specimen configurations of similar size. The limiting values for the major controlled parameters were specified for design purposes to have the ranges of values shown in Table 5.

These ranges of values provide up to a sixfold increase in the motion rate and in the forcing rate, up to a one hundredfold increase in the preload force or in the cyclic force, and up to approximately a thirty-fold increase in the motion amplitude as compared to "normal" operating conditions which lead to either fretting-wear or wear in the selected test specimen. However, the highest speed, largest force, and maximum motion will probably never be combined due to the large power requirement and excessive heating of the specimen.

TABLE 5. RANGES OF IMPORTANT PARAMETERS ESTABLISHED AS REQUIREMENTS FOR THE WEAR/FRETTING-WEAR TESTING MACHINE

Parameter Description	Minimum Limiting Value	Maximum Limiting Value
Static preload force, lb	50	5000
Amplitude (half the peak-to-peak value) of cyclic force about the mean, lb	50*	2500
Peak force (static preload plus cyclic force), lb	50	5000
Frequency of cyclic force, cpm	600*	3600
Minimum peak-to-peak magnitude of relative cyclic motion (determined for position with largest motion), deg	.1**	45**
Frequency of cyclic motion, cpm	600	3600
<p>* These are minimum values for the cyclic force when cyclic forcing is employed. For constant preload tests, the cyclic force parameters are zero.</p> <p>** 0.1 degree corresponds to a peak-to-peak relative motion of about .002 inch and 45 degrees to .84 inch for the 2.155-inch-diameter specimen.</p>		

In addition to having the capability for producing the specified controlled ranges for each of these parameters, the testing machine must embody means for accurately measuring the forces, motions, cycles, and torques applied to the test piece. Specifically, the testing machine, which will not be constructed until a later time, shall include appropriate transducers and instrumentation to:

- (1) Measure the oscillatory sliding motion during the test to an accuracy of $\pm .001$ inch or $\pm 5\%$ of the true value, whichever is less, for peak-to-peak motions of .002 inch to .070 inch.
- (2) Measure the oscillatory sliding motion during the test to $\pm \frac{1}{4}$ degree for motions larger than .070 inch peak-to-peak.

- (3) Measure the cyclic force continuously during the test with an accuracy of $\pm 5\%$ of the true force or ± 25 lb, whichever is less, for loads up to 5000 lb.
- (4) Measure and adjust the preload force while testing to an accuracy of $\pm 5\%$ of the true force or ± 25 lb, whichever is less, for loads up to 5000 lb.
- (5) Indicate the number of oscillations or cycles to the nearest 100 cycles up to 10 million cycles.
- (6) Measure the torque required to oscillate the test piece during the testing process to an accuracy of $\pm 5\%$ of the true value up to 10,000 in.-lb.
- (7) Monitor the temperature of the test piece at appropriate positions near the sliding surfaces to an accuracy of ± 5 degrees Fahrenheit or $\pm 5\%$, whichever is greater.

DESIGN OF FRETTING-WEAR TESTING MACHINE

SPECIMEN CONFIGURATIONS

The configuration of the cyclic servo support bearing, selected as the test specimen for the proposed testing program, strongly influenced the design of the testing machine. However, it was not the only specimen geometry considered. It was recognized that cylindrical specimens have certain advantages, including not only easier assessment of contact pressures and motions, but lower manufacturing costs. Since the rotational axis of planar motion of a sphere and planar motion of the same size cylinder can easily be made to coincide, the same basic motion and force-producing techniques can be employed with either geometry.

With any test specimen geometry it is required that the contacting surfaces be capable of disassembly without permanent damage to the parts or noticeable disturbance to the contacting surfaces. For a spherical specimen, this can be accomplished by supporting the sphere in a two-piece spherical socket, or bushing, which is cut across a diametral plane. Conformance to the spherical ball element within satisfactory tolerance limits requires that the socket contact only a part of the surface of the sphere. The spherical contact zone may be reduced to only a narrow supporting band or ring on either side of a diametral plane. For a cylinder, satisfactory disassembly can be accomplished in a similar manner by using a cylindrical bushing cut along a diametral plane. It is also possible to use uncut cylindrical bushings, but there is a much greater likelihood of disturbing the contact areas when disassembly is attempted. The cylindrical specimen can be gripped at the ends and a single bushing used, or it can be gripped near the center of its length and bushing used at each end. Thus, a variety of specimen configurations is possible with spherical and cylindrical geometries, some of which may be commercially available.

The spherical bearing selected as the specimen for this investigation is produced to rigid specifications and is commercially available. The bearing consists of a spherical element and a two-piece separable bushing set is shown in Figure 2. Attachment to the spherical element can be made using the large central hole through the sphere.

METHOD SELECTED FOR PRODUCING FORCES AND MOTIONS

The forces and motions for the wear and fretting-wear phenomena could be supplied to the chosen spherical specimen in any of several ways. The bushing could be retained in a stationary housing with the loads and motions applied through actions on the ball element. This would simulate the actual helicopter application but load the two bushing-ball interfaces differently, since the load would react primarily against one bushing for a "pull" load and on the opposite bushing for a "push" load.

For a second possibility, the housing could be rotated and the load applied through the ball. This would have the unequal loading disadvantage, plus the requirement that the load on the ball be supported by the bearings that permit the housing to rotate. A third possibility would be to both load and move the bushing housing. This would again have the disadvantage of requiring support of the applied load through auxiliary bearings.

A fourth possibility, which eliminates the disadvantages of the first three arrangements suggested, would be to rotate the ball and load one bushing against the ball, letting the other bushing provide the reaction against a stationary base. In this manner, both bushings would be forced against the spherical element with the same axial force and would be subjected to the same sliding motion at the same time, assuming the bushings remain parallel to each other. This fourth arrangement was adopted for the testing machine. Two tests can be conducted with one specimen using this method of applying the force and motion: one test at each of the ball-ring interfaces.

FORCE CONTROL AND MEASUREMENT

The total normal force is composed of a preload and a superposed cyclic load which must be independently adjustable to any desired value within the specified design range. Using the chosen force application technique described in the last section, the specimen is loaded by pressing one bushing against the ball element. It was found experimentally that the parts involved are relatively rigid, so that very little axial displacement of the loaded bushing is required to produce the maximum design load on the bearing. An axial force of 5000 lb caused one bushing to be displaced axially only a few thousandths of an inch relative to the other bushing. Therefore, a lever mechanism with a 10 to 1 mechanical advantage was considered to be an adequate means for loading the ball. Such a lever acts as a motion reducer and a force amplifier at the specimen loading ram positioned inside the fulcrum and connected to the lever by a flexure plate. The short arm of the lever was attached at the fulcrum to the rigid machine frame by means of additional flexure plates. The flexures have the advantage of eliminating bearings at these relatively high load points and provide a more dimensionally stable design by eliminating points of wear in the testing machine itself.

The displacement of the long arm of the lever in the appropriate direction imposes a load on the ram and on the test specimen. By adjusting the position of the lever, a preload may be applied to the specimen, and by oscillating the end of the lever about this position a cyclic force is superposed. Thus, a variable displacement device mounted on an adjustable base was selected for the loading function. A commercially available variable-throw eccentric mounted on a sliding base was designed to provide the force control system. The eccentric and cyclic loading can be adjusted only when the eccentric is not rotating, but the preload can be adjusted while running the test.

The forces acting on the specimen are measured by a compact, flat-type load cell mounted in series with the ram between the loading lever and the upper loaded bushing of the specimen assembly. The time variation of the force acting on the specimen can be monitored continuously during the test. This provides information on both the preload and the cyclic load, and allows an observer to note any changes that may occur during a test.

TWO MOTION RANGES

Attempts to provide the full range of motion needed for both fretting and fretting-wear with a single mechanism resulted in a crank throw that was so small on the low end of the range that it could not be set with the required degree of accuracy or was so large on the other end that the dimensions of the crank mechanism and the dynamic forces became excessive.

An acceptable compromise was reached by providing one mechanism for the fretting-wear motions and another for wear. The range of motions for fretting, up to .070 inch, was provided by a linkage consisting of a very rigid long follower attached to the ball element, driven by a variable throw eccentric. The long follower provided the motion amplification needed but imposed an elastic deflection problem that required a very rigid structural configuration. With too much elastic deflection of the follower arm, the motion desired for testing would not be available at the contacting surfaces and would not be controllable or consistent throughout the duration of the test.

The wear motion range overlaps part of the fretting-wear motion range and is produced by a linkage with a much shorter follower link and a longer connecting link. The short follower permits relatively large rotational motions for moderate crank throws, and the long connecting rod allows the pressure angle to be maintained within an acceptable range. However, this linkage cannot be operated from the same driver position required for the fretting-wear motion. This does not require a different variable throw eccentric, but it does require that the same eccentric used for fretting-wear be relocated to the new driver position when wear tests are run. It also requires another bearing-mounted shaft to minimize alignment problems when remounting the variable throw eccentric.

MOTION MEASUREMENT

For the range of sliding motion established for fretting, .002 inch to .070 inch peak-to-peak, motion measurements must be made very close to the sliding interface to eliminate errors due to deflections or clearances in intermediate load-carrying parts. For example, the motions of the eccentric crank, or of points on the stem attached to the spherical element, are much larger than the interface motions and could be readily measured, but any clearances in the connecting bearings or deflections

in the stem would be included in such a measurement. To avoid this difficulty, an auxiliary reference plane was clamped directly to and around the center of the spherical element between the bushings. This provided a slight magnification of the motion of the surface and established a reference position for the ball which was not subjected to the torque acting on the sphere. Thus the angular displacement of this auxiliary reference plane relative to the bushings is a direct measure of the sliding motion at the test specimen interface.

The amplitude of the angular displacement of the reference plane can be measured with a noncontacting, displacement sensitive probe attached to the bushings and directed at the surface of the auxiliary reference plane. One probe should be located at the region of maximum motion on one side of the test specimen, with a second probe located 180° away to ascertain that the sliding behavior is symmetrical. A second set of probes should be mounted on one bushing and referenced to the other bushing to measure the axial or translational motion that occurs between the bushings and the ball, and to indicate any tipping that may occur between the bushings as, for example, could occur if one bushing rotated with the sphere. This set of probes is also useful for monitoring the amount of wear as a function of time, since the mean axial displacement as a function of time is a direct measure of total wear.

For the larger amplitude motions encountered with wear, the auxiliary reference plane must not be clamped to the ball because it would interfere with the bushings. In its place a pointer can be attached to the wear stem such that it indicates the magnitude of angular displacement on a stationary scale attached to the frame. The scale can be calibrated in inches of relative motion or in degrees of angular displacement and can be read with the aid of a strobe light set to the oscillating frequency of the motion. With a magnification factor of approximately 5 to 1, the peak-to-peak motions should be readable to $\pm \frac{1}{4}$ degree (equivalent to about $\pm .005$ inch) without great difficulty. This accuracy is acceptable for the relatively large motion amplitudes associated with wear.

TORQUE MEASUREMENT

The torque measurement for fretting-wear can be related to the strain in the stem attached to the ball. The strain in the stem can be measured by bonding a dynamic strain gage to the stem near its junction with the spherical member of the test specimen. Since this is the region of greatest strain and least rigid-body motion, the leads to the strain gages may be brought out without the use of slip-rings. By mounting gages 180° apart in the alternating high tension and compression regions on the stem, it is possible to increase the output of the bridge circuit. However, even by amplifying the signal from the low strain tests, it was not possible to obtain satisfactory signals for the entire torque range of 30 to 10,000 in-lb.

To improve the torque measurement sensitivity in the low strain tests, the range was separated into two parts. One stem was modified by cutting out material behind the strain gages and leaving only a thin strip of metal for attaching the gages. This created a higher strain for the low torques but created higher stresses which could not be tolerated for high torque loading. Thus, a modified fretting-wear stem was designed for torque loads up to 3500 in-lb. The overlap was provided to insure adequate range for those tests where the friction and torque might change substantially during the testing operation.

The torque measurement for wear also employs the strain measurement technique with gages mounted on the wear stem.

BALANCING OF MOVING PARTS

The adjustable throw, eccentric cranks are balanced by removing a mass of material from the cylindrical insert of the eccentric at a location axially in line with and behind the projecting crank-pin located on the insert. The mass removed must be equivalent to the sum of the mass of the pin plus the bearing mounted on the pin plus the part of the end of the attached link that can be considered to move with the pin. Thus, when the cylindrical insert is turned in the eccentric housing to provide a different throw, its effective center of mass does not change, and the balance of the eccentric with respect to the axis of rotation is not disturbed.

The reciprocating parts, including the ball, stems connected to the ball and associated links, were not balanced as easily as the rotating components. Several techniques were considered, from which evolved the idea of having an equivalent, but counterrotating dummy linkage in the plane of the test linkage. However, since the dummy mechanism consisted of essentially the same members as those required for the test mechanism, it was decided to make the balancing linkage even more useful by converting it to a second testing station, thereby permitting two specimens to be run at the same time with only a small increase in cost and effort.

A gear arrangement was deemed necessary to provide this counter-rotational motion since the crossed-belt concept or the technique of driving with the back side of a two-sided belt or chain did not produce a good design for long term testing. Difficulty in obtaining a suitable gearbox, with a 1:1 gear ratio and parallel input-output shafts that were capable of transmitting the necessary horsepower at the speed desired, led to the idea of rotating the two motion mechanisms in the same rotational direction but 180° out of phase. This arrangement eliminated the need for the gearbox.

The phase relationship between the motion cranks is assured by the use of timing belts to drive both units. The cyclic force mechanisms are also driven by means of timing belts connected to the motion drives.

This permits positive phase adjustments between the motion and cyclic force parameters.

TEMPERATURE MEASUREMENT

The temperature of the specimen surfaces at the fretting-wear interface will vary from point to point due to differences in motion amplitude, sliding rate, normal pressure, heat removal rate, friction characteristics, and other factors. It is therefore desirable to measure a characteristic temperature at some representative location, ideally at the contacting surface interface, but more practically some distance away from this surface. Other considerations being equal, it is expected that the maximum temperature will occur in the region of greatest motion and will be the most characteristic temperature for test purposes.

The bushings are more directly accessible and the temperature of these elements can be measured by thermocouples either embedded in a hole drilled into the bushing or placed in contact with the surface of the bushing. The drilling process would be much more difficult since the bushings are hardened, and the drilling probably would employ an EDM (electrical discharge machining) process to prevent extensive changes in the bushing material around the hole. It is felt that embedding a thermocouple in several samples to compare the surface and interior readings would be a more reasonable approach than embedding thermocouples in every bushing. The surface thermocouple can be placed on the circumferential surface of the bushings and held in contact with the bushing clamps.

It is also planned to use a radiation type temperature-indicating device directed at the ball surface to indicate its temperature, especially when larger amplitude motions expose portions of the contact surface.

FACILITIES FOR COOLING THE SPECIMEN

In the event it becomes desirable to cool the specimens during a test to maintain a specific temperature for accelerated tests, or to evaluate the effect of a parameter without an associated temperature rise, the specimen can be cooled by directing air across the exposed part of the spherical surface and the exposed face of the bushing. The angular displacement of the sphere exposes part of the contact surface during each cycle, and the air would increase heat removal from this region. Also, heat conducted to the supporting parts would be removed more rapidly. The air must not be used in such a manner that it removes debris from the nonexposed interface regions.

The insides of the stems have been so designed that air can be brought into the internal part of the sphere and directed against the inside of the sphere by means of grooves running axially along the shank of the fretting-wear and wear stems. The air is collected in a relief groove

around the stem inside the ball and is exhausted through ports on opposite sides of the stem.

DAMAGE MEASUREMENT AND FAILURE DETECTION

A clear definition of failure by wear or fretting-wear is often elusive. Unlike failure modes such as fatigue or ductile rupture, where failure is marked by a sudden catastrophic event, wear and fretting-wear build gradually toward failure by loss of function of the affected part, and the definition of the moment of failure is difficult. For example, bearings used in Army helicopters have specified limits for looseness (play) and roughness, but the cyclic servo support bearing can be adjusted to eliminate looseness, so a specified amount of play would not be meaningful in defining failure.

In designing the testing machine described here, three techniques of wear and/or fretting-wear damage measurement have been implemented:

- (1) Measurement of mean axial displacement of one cyclic servo bushing ring with respect to the other ring as a function of operating time.
- (2) Measurement of the cyclic torque amplitude required to move the ball through a specified angular displacement with respect to the bushing rings, as a function of time.
- (3) Periodic disassembly inspection and profilometer or microscopic measurement of the wear scar profile.

Failure definition can be established only after some testing experience has been evaluated and related to operational field failures. During preliminary testing it will be necessary to define failure in terms of a limiting value of mean axial displacement and/or a limiting value of drive torque increase as the test progresses. The transducers to make these measurements have been described in earlier sections on torque measurement and motion amplitude measurement. It would be proposed to relate these measurements to visual, microscopic, and profilometric data periodically obtained for the test specimens and similar data obtained from actual cyclic servo support bearings recovered from the field as maintenance discards. Preliminary experimental evaluations of virgin specimens versus field failures indicate that an increase in driving torque of more than 50% may be expected by the time wear or fretting-wear failure occurs. While final development of these damage measurements and failure detection techniques must await construction of the testing machine, it now appears that these functions can be satisfactorily achieved by carefully integrating the results of all three measurement techniques proposed.

SUMMARY AND CONCLUSIONS

Primary objectives of the work reported here included the definition of potentially useful failure prediction models for the wear and fretting-wear failure modes, definition of parameters of primary importance, incorporation of accelerated testing concepts in the prediction models and in the design of a meaningful testing program, and the design of a special testing machine capable of both real-time and accelerated wear and fretting-wear tests.

Three prediction models are recommended for experimental investigation: the adhesive/abrasive model, the zero wear model, and the linear cumulative damage model. Of these, the generalized linear cumulative damage model requires the least specific information about the physical process of wear or fretting-wear and is most readily adaptable to accelerated testing. Preliminary estimates indicate that effective test-time acceleration factors exceeding five should be achievable.

An extensive literature survey augmented by independent research indicates that, while there are scores of factors that may influence wear and fretting-wear processes, there are twelve factors of primary interest. These factors must be controlled and/or measured in the testing program, and they must be provided for in the design of the testing equipment.

In the design of a testing program it has been possible to establish baseline control conditions by studying the operational loads, motions and other parameters associated with the service environment of a UH-1H cyclic servo support bearing which was chosen as the "specimen" to be tested. Using these baseline conditions as a comparison standard, a testing program has been designed to evaluate constants for the proposed prediction models and to test the validity of the linear cumulative damage hypotheses as a tool for meaningful accelerated tests for wear and fretting-wear damage. A preliminary feasibility study has been outlined which would require about 30 cyclic servo support bearings and about four months of testing if two double-test-bed machines were available for the program. An extended testing program is also outlined, to be executed only if the preliminary feasibility study program produces positive results. The extended testing program would require approximately 300 cyclic servo support bearings and nearly three years to complete.

The design of a special testing machine has been completed, and engineering drawings have been prepared for its construction. While versatility in specimen configuration has been incorporated in the design, the specimen of choice is the UH-1H cyclic servo support bearing. The machine is capable of producing either wear or fretting-wear under a wide range of conditions. Actual service conditions of all the important parameters may be readily simulated, controlled and measured. Provisions for extensive changes in loading, motion and frequency parameters have been built in to accommodate a wide range of accelerated test conditions.

The machine has been designed specifically to execute the testing program outlined, but has been so conceived that it may be extended to many other investigations in the future which might involve different material combinations, finishes, platings, coatings, lubricants, and geometrical configurations in wear or fretting-wear environments.

The importance of wear and fretting-wear as failure modes is clearly documented. Wear or fretting-wear may, in some instances, result only in a loss of proper fit which simply requires replacement of a worn part. In other cases, these phenomena may result in loss of function, seizure, or loss in control of a critical system to produce a catastrophic failure. Because of the importance of these failure modes, it is recommended that the machine construction and testing program outlined in this report be executed. It is recommended that two double-test-bed machines be constructed in accordance with the engineering drawings, and that the proposed testing program be executed using these machines. The potential improvements in wear prediction through accelerated testing, and long-range economic benefits, as well as improved mechanical reliability, are thought to justify such an investment.

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LIST OF SYMBOLS

d_w	depth of adhesive/abrasive wear, in.
f_θ	normal operating frequency of angular motion, cpm
k_w	experimental adhesive/abrasive wear constant, in. ² /lb
L_1, L_2	life at a specific stressor level, cycles or hr
L_{hi}	life at a high stressor value, cycles or hr
L_{op}	life at operating stressor level, cycles or hr
L_p	time to failure with the load P fluctuating at its standard reference frequency, in wear test series in which the operational load amplitude is the stressor, hr
L_{ps}	time to failure under static load P , in wear test series in which the operational load amplitude is the stressor, hr
L_s	distance of sliding, ft or in.
L_θ	time to failure with load P fluctuating at its standard reference frequency, in wear test with operational frequency of angular motion as the stressor, hr
$L_{\theta s}$	time to failure under static load P , in wear test series in which the operational frequency of angular motion is the stressor, hr
N	maximum number of passes for zero wear
P	normal operating amplitude of axial load, lb
P_{hi}	force used in accelerated test, lb
P_m	mean nominal contact pressure, psi
P_{op}	normal operating force, lb

LIST OF SYMBOLS - Continued

S_1, S_2	stressor levels
$T_{0.25}$	time to failure under static baseline conditions of Table 1 but with load of $5P$, <u>after</u> prior operation at static baseline conditions of Table 1 with load of P for a duration of $0.25 L_{Ps}$, hr or day
$T_{0.75}$	time to failure under static baseline conditions of Table 1 but with load of $5P$, <u>after</u> prior operation at static baseline conditions of Table 1 with load of P for a duration of $0.75 L_{Ps}$, hr or day
$t_{0.25}$	time to failure under dynamic baseline conditions of Table 1 but with load of $5P$, <u>after</u> prior operation at dynamic baseline conditions of Table 1 with load of P for a duration of $0.25 L_P$, hr or day
$t_{0.50}$	time to failure under dynamic baseline conditions of Table 1 but with load of $5P$, <u>after</u> prior operation at dynamic baseline conditions of Table 1 with load of P for a duration of $0.50 L_P$, hr or day
$t_{0.75}$	time to failure under dynamic baseline conditions of Table 1 but with load of $5P$, <u>after</u> prior operation at dynamic baseline conditions of Table 1 with load of P for a duration of $0.75 L_P$, hr or day
$t'_{0.25}$	time to failure under dynamic baseline conditions of Table 1 but with cyclic motion of $5f_0$, <u>after</u> prior operation at dynamic baseline conditions of Table 1 with cyclic motion frequency of f_0 for a duration of $0.25 L_0$, hr or day

LIST OF SYMBOLS - Continued

$t'_{0.50}$	time to failure under dynamic baseline conditions of Table 1 but with cyclic motion of $5f_a$, <u>after</u> prior operation at dynamic baseline conditions of Table 1 with cyclic motion frequency of f_a for a duration of $0.50 L_a$, hr or day
$t'_{0.75}$	time to failure under dynamic baseline conditions of Table 1 but with cyclic motion of $5f_a$, <u>after</u> prior operation at dynamic baseline conditions of Table 1 with cyclic motion frequency of f_a for a duration of $0.75 L_a$, hr or day
t_1, t_2, t_3	time of operation at stressor level 1, 2, etc., hr
α	fraction of life at which component is operated at stressor level S_1
β	fraction of life at which component is operated at stressor level S_2
γ_r	experimental constant
σ_{yp}	yield point strength of the material, psi
τ_{max}	maximum shearing stress, psi
τ_{yp}	shear yield point of the material, psi